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RESEARCH MEMORANDUM

for the

Bureau of Aeronautics, Department of the Navy

EFFECT OF AIR COOLING OF TURBINE DISK ON POWER AND EFFICIENCY OF TURBINE FROM TURBO ENGINEERING CORPORATION TT13-18 TURBOSUPERCHARGER

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NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS

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SUMMARY

An investigation was conducted to determine the effect of turbine-disk cooling with air on the efficiency and the power output of the radial-flow turbine from the Turbo Engineering Corporation TT13-18 turbosupercharger. The turbine was operated at a constant range of ratios of turbine-inlet total pressure to turbine-outlet static pressure of 1.5 and 2.0, turbine-inlet total pressure of 30 inches mercury absolute, turbine-inlet total temperature of 1200° to 2000° R, and rotor speeds of 6000 to 22,000 rpm.

Over the normal operating range of the turbine, varying the corrected cooling-air weight flow from approximately 0.30 to 0.75 pound per second produced no measurable effect on the corrected turbine shaft horsepower or the turbine shaft adiabatic efficiency. Varying the turbine-inlet total temperature from 1200° to 2000° R caused no measurable change in the corrected cooling-air weight flow.

Calculations indicated that the cooling-air pumping power in the disk passages was small and was within the limits of the accuracy of the power measurements. For high turbine power output, the power loss to the compressor for compressing the cooling air was approximately 3 percent of the total turbine shaft horsepower.

INTRODUCTION

An investigation of the radial-flow turbine from the Turbo Engineering Corporation TT13-18 turbosupercharger has been conducted

at the NACA Lewis laboratory, at the request of the Bureau of Aeronautics, Department of the Navy, to determine the effects of air cooling the turbine disk on the efficiency and the power output of the turbine. The results of other phases of this investigation are reported in references 1 and 2.

Air cooling of the turbine disk was provided to increase the life of the rotor and to increase the permissible inlet gas temperatures. The cooling passages were located at the periphery of the disk at the blade roots. Reference 3 shows that air cooling of the periphery of the disk and thereby cooling of the blade by conduction of heat from the blades to the cooled disk permits only small increases in the inlet total temperature; the principal advantage of this type of cooling therefore appears to be that either the allowable stress in the disk or the life of the disk may be increased.

For the present investigation, turbine operating conditions ranged from speeds of 6000 to 22,000 rpm, ratios of inlet total pressure to outlet static pressure of 1.5 and 2.0 at an inlet total pressure of 30 inches of mercury absolute, and inlet total temperatures from 1200° to 2000° R. The cooling-air weight flow was varied from approximately 0.30 to 0.75 pound per second for each set of turbine operating conditions.

SYMBOLS

The following symbols are used in the calculations and on the figures:

A area, square feet

g acceleration due to gravity, 32.174 feet per second per second

Δh isentropic enthalpy change, Btu per pound

N speed of rotation, rpm

P total pressure, pounds per square foot

p static pressure, pounds per square foot

R gas constant, foot-pounds per pound per °R

T total temperature, °R

V axial velocity, feet per second

W gas weight flow, pounds per second

γ ratio of specific heats

δ ratio of inlet total pressure to NACA standard sea-level pressure, $(P_i/2116.8 \text{ lb/sq ft})$

η_s turbine-shaft adiabatic efficiency (ratio of measured shaft power to theoretically available power based on total inlet temperature and pressure and outlet static pressure)

θ ratio of inlet total temperature to NACA standard sea-level temperature, $(T_i/518.4^\circ \text{ R})$

ρ mass density of driving fluid, slugs per cubic foot

Subscripts:

a cooling air

i inlet

o outlet

t turbine

APPARATUS AND PROCEDURE

The apparatus used for this investigation is the same as that used in reference 2 with the exception of modifications in the exhaust duct to facilitate collection of the cooling air as it leaves the disk.

Turbine

The TT13-18 turbine is designed for radial flow, in which the driving fluid enters the wheel radially from a double, tangential-inlet, radial-flow nozzle box and discharges axially into the exhaust pipe.

The turbine rotor and its cooling-air passages are shown in figures 1 to 3. The cooling air enters the turbine wheel on the shaft side of the disk through 51 drilled passages (fig. 1) and is discharged from the downstream face of the wheel through the 17 holes

shown in figure 2. Figure 3 is a cross-sectional view of these passages showing how the 51 inlet passages combine in groups of three into the 17 larger outlet passages, which lead to the downstream face of the turbine wheel; each of these outlet passages lies approximately parallel to the root of a turbine blade. Figure 3 also shows that each of the 17 outlet passages was threaded to give more surface area for heat transfer to the cooling air. In order to reduce the pressure loss in the inlet to the cooling-air passages, the inlet cooling-air passages were drilled at an angle such that the projection of a cooling-air passage on a tangential plane makes an angle of approximately 45° with the axis of rotation. The outlet cooling-air passages were also drilled at such an angle that the projection of a cooling-air passage on a tangential plane makes an angle of approximately 45° with the axis of rotation in order to recover a portion of the tangential momentum of the cooling air leaving the rotor.

Cooling-Air System

In the turbosupercharger the turbine cooling air is taken from the outlet of the first-stage compressor. Inasmuch as the compressors were removed from the unit, cooling air had to be externally supplied. In order to supply this air, the compressor casing was retained and the compressor inlet and one of the two compressor outlets were closed off. The remaining compressor outlet was connected to the laboratory 40-pound-per-square-inch air system. The cooling-air flow path is indicated by arrows in figure 4.

The cooling-air inlet chamber could not be made leakproof because of the necessary running clearance required between the shaft side of the turbine wheel and the nozzle-box support. The turbine was originally equipped with a labyrinth seal (fig. 3) to minimize the leakage. The cold clearance of this seal is approximately 0.009 inch as measured. With this clearance of 0.009 inch, the labyrinth leakage area is less than 0.06 of the total cooling-air outlet passage area in the turbine wheel. The running clearance was indeterminate and probably deviated somewhat from the cold clearance. Leakage occurred between the cooling-air inlet chamber and the turbine nozzle box, the direction of leakage depending on which chamber was operating at the higher pressure. The cooling air at the outlet of the cooling passages in the turbine disk discharges into a conical collector. Clearance between this collector and the wheel was held at 0.025 to 0.040 inch.

Instrumentation

Hot-gas measurements. - Air flow to the hot-gas producer was measured with a conventional thin-plate orifice in accordance with A.S.M.E. specifications. Fuel flow was measured with a calibrated rotameter.

Turbine-inlet total temperature was taken as the average of four quadruple-shielded chromel-alumel thermocouples in the turbine-nozzle-box-inlet pipes, two thermocouples in each pipe as shown in figure 4. Turbine-inlet static pressure was read as the average of eight static-pressure taps, four each located in the same plane and 90° apart in each of the two turbine-nozzle-box-inlet pipes. The turbine-outlet static pressure was taken as the average of five static-pressure taps equidistantly spaced around the outer periphery of the outlet pipe approximately 5 inches downstream of the trailing edge of the turbine blades.

The inner part of the cylindrical exhaust pipe shown in figure 1 of reference 2 was removed and replaced with the conical cooling-air collector shown herein in figure 4. With this configuration, the turbine-outlet gas was subjected to an abrupt expansion, which caused discrepancies in the turbine-outlet static-pressure measurements. These discrepancies led to variations, up to 2 percent over the range investigated, in performance results between these data and those reported in reference 2. The variations were consistent, however, and did not affect the value of the results of this investigation.

Cooling-air measurements. - The cooling-air-inlet static pressure was taken as the average of three static-pressure taps on the wall of the annulus approximately 5 inches upstream of the turbine wheel. Cooling-air-inlet total temperature was taken as the average of three iron-constantan thermocouples located at approximately the same place as the inlet static-pressure taps.

Four static-pressure taps were located in the cooling-air collector approximately 1 inch downstream of the cooling-air passage outlets in the turbine wheel. The average of these instrument readings was taken as the cooling-air-outlet static pressure.

Turbine cooling-air weight flow was measured by a standard A.S.M.E. orifice similar to that used for measuring the combustion air flow to the hot-gas producer. The amount of air leakage around the disk was not determined.

Torque and speed measurements. - Torque measurements were made with an NACA balanced-diaphragm torquemeter operating with a high-speed dynamometer. Turbine speed was measured with an electric chronometric tachometer that was driven by a tachometer generator turning at one-tenth turbine speed.

Accuracy. - The accuracy obtained with the measurements were within the following limits:

Turbine-gas flow, lb/sec	±0.06
Cooling-air flow, lb/sec	±0.02
Static pressure, in. Hg	±0.05
Chromel-alumel thermocouples, °R	±5
Iron-constantan thermocouples, °R	±2
Torque, ft-lb	±0.2
Turbine speed, rpm	±10

Procedure

The turbine was operated at a constant inlet total pressure over a range of pressure ratios, inlet total temperatures, and rotor speeds. For each operating condition, the cooling-air weight flow was varied from approximately 0.30 to 0.75 pound per second. Turbine operating procedure was similar to that in reference 2, except for the variable cooling-air flow used in this investigation. The following table shows the operating range:

Turbine pressure ratio	Turbine-inlet total temperature (°R)	Turbine rotor speed, rpm
1.5	1200	6000 - 22,000
1.5	1600	6000 - 22,000
2.0	1200	12,000
2.0	1400	12,000
2.0	1800	12,000
2.0	2000	12,000

METHOD OF CALCULATION

Turbine-inlet total pressure was calculated from measured turbine-inlet static pressure, weight flow, and annulus area. The continuity equation is used to find the velocity.

$$W_t = g\rho_{i,t} A_{i,t} V_{i,t}$$

Turbine-inlet total pressure is defined as

$$P_{i,t} = p_{i,t} + \frac{\rho_{i,t} V_{i,t}^{2}}{2}$$

Turbine efficiency was taken as the ratio of measured shaft power to the theoretically available power in the driving fluid based on the ratio of turbine-inlet total pressure to turbine-outlet static pressure. The theoretically available power was determined from turbine weight flow and a chart (fig. 8, reference 4), which is a plot of ideal turbine work against turbine pressure ratio for a range of turbine-inlet total temperatures and fuel-air ratios. This chart is applicable only to operation with a fuel having a hydrogen-carbon ratio of 0.189. Because the accuracy of the chart decreases at the low pressure ratios, a check was made by the equation

$$\Delta h_t = \frac{\gamma_{i,t}}{\gamma_{i,t}-1} RT_{i,t} \left[1 - \left(\frac{p_{o,t}}{P_{i,t}} \right)^{\frac{\gamma_{i,t}-1}{\gamma_{i,t}}} \right]$$

for ideal turbine work (reference 5). The ratio of specific heats $\gamma_{i,t}$ was obtained from figure 7 of reference 4, a chart showing γ for a range of turbine-inlet total temperatures and fuel-air ratios. Use of this alternate method showed the chart to be accurate within ±0.2 percent in this range of turbine pressure ratios.

Shaft power was directly calculated from the observed torquemeter readings. Bearing losses were not considered because they are constant for any given set of conditions.

RESULTS AND DISCUSSION

The corrected cooling-air weight flow supplied for a range of turbine speeds from 3970 to 18,900 rpm at constant ratios of inlet

total pressure to outlet static pressure and turbine-inlet total pressure and temperature is shown in figure 5. For these conditions the corrected cooling-air weight flow increased with turbine speed for a given cooling-air pressure ratio, which is produced by centrifugal forces in the cooling-air passages. The centrifugal force on the cold gas in the long inlet cooling-air passages is greater than the centrifugal force on the hot gas in the outlet cooling-air passages, an effect that increases with speed. For example, the corrected cooling-air weight flow increased from 0.57 to 0.72 pound per second at a cooling-air pressure ratio of approximately 1.5 over a range of turbine speeds from 3970 to 18,900 rpm. The cooling-air weight flow represents the total amount of cooling air supplied to the turbine and is the sum of the cooling air that passes through the cooling-air passages and that which flows through the cooling-air labyrinth seal. With the 0.009-inch clearance for the labyrinth seal, the cooling air that passed through the rotor should not significantly differ from the measured cooling-air weight flow.

Effects of Disk Cooling on Performance

Corrected turbine shaft power plotted against corrected cooling-air weight flow is shown in figure 6. Over a range of corrected turbine speeds at a constant ratio of inlet total pressure to outlet static pressure and inlet total pressure and temperature, there is no measurable effect on turbine power output with a change in corrected cooling-air weight flow from 0.35 to 0.75 pound per second. The turbine-shaft adiabatic efficiency did not vary with corrected cooling-air weight flow, as shown in figure 7. The change in turbine power due to leakage of cooling air through the cooling-air labyrinth seal is so small that it may be neglected.

The effect of turbine-inlet total temperature on the corrected cooling-air weight flow is shown in figure 8. There appears to be no change in cooling-air weight flow over a range of temperatures from 1200° to 2000° R.

Cooling-Air Power Losses

Pumping losses in cooling-air passages. - The cooling-air passages in the disk discharge the air at a radius from the center line of the shaft greater than the radius of the inlet ports. For low cooling-air weight flows and high rotor speeds, a small amount of energy is imparted to the cooling air by the turbine wheel because these passages act as a compressor. For high cooling-air weight

flows and low rotor speeds, however, the tangential component of the discharged cooling air may exceed the whirl velocity of the rotor at the radius of the cooling-air outlet ports, in which case a small amount of work will be done on the wheel by the cooling air.

Calculations of this pumping power were made over a wide range of cooling-air weight flows and the results indicate the pumping power to be within the limits of accuracy of the power measurements.

Compressor power loss. - When the turbine is operating as a component of a turbosupercharger, part of its power must go to the compressor for compressing the air that is supplied for disk cooling.

Compressor power required for compressing the cooling air plotted against cooling-air pressure ratio is shown in figure 9. The lines of constant cooling-air weight flow represent the approximate limits for this investigation. At a cooling-air pressure ratio of 1.5 and a maximum cooling-air weight flow of 0.70 pound per second, approximately 20 horsepower are required to compress the air, with an assumed compressor efficiency of 0.75 and compressor-inlet total temperature of 518.4° R. For comparison, a two-stage turbosupercharger was assumed to be operating with this turbine at a ratio of turbine-inlet total pressure to turbine-outlet static pressure of 3.0, a turbine-inlet total pressure of 50 inches of mercury absolute, and a turbine-inlet total temperature of 1600° R. For these conditions reference 2 shows that the turbine power output will be approximately 600 horsepower. A 20-horsepower loss for compressing the cooling air would be an over-all loss of 3 percent of the total output of the turbine.

SUMMARY OF RESULTS

An investigation of the effects of air cooling of the disk on the power and the efficiency of the turbine from the Turbo Engineering Corporation TT13-18 turbosupercharger yielded the following results:

1. Over the normal operating range of the turbine, corrected cooling-air weight flows from 0.30 to 0.75 pound per second had no measurable effect on the corrected turbine-shaft power output or the turbine-shaft adiabatic efficiency.

2. With all other turbine operating conditions held constant, a change in turbine-inlet total temperature from 1200° to 2000° R had no measurable effect on the corrected cooling-air weight flow.

3. At a cooling-air pressure ratio of 1.5, a ratio of turbine-inlet total pressure to turbine-outlet static pressure of 2.0, and a turbine-inlet total temperature of 1600° R, the corrected cooling-air weight flow varied from 0.57 pound per second at a corrected turbine speed of 3970 rpm to 0.72 pound per second at a corrected turbine speed of 18,900 rpm.

4. At a compressor pressure ratio of 1.5 and along a compressor performance line of constant efficiency equal to 0.75, the power required to compress 0.70 pound of cooling air per second was 20 horsepower. With the turbine operating at a ratio of inlet total pressure to outlet static pressure of 3.0, an inlet total pressure of 50 inches of mercury absolute, and an inlet total temperature of 1600° R, the power loss to the compressor for compressing 0.70 pound of cooling air per second at a pressure ratio of 1.5 would be 3 percent of the turbine output.

Lewis Flight Propulsion Laboratory,
National Advisory Committee for Aeronautics,
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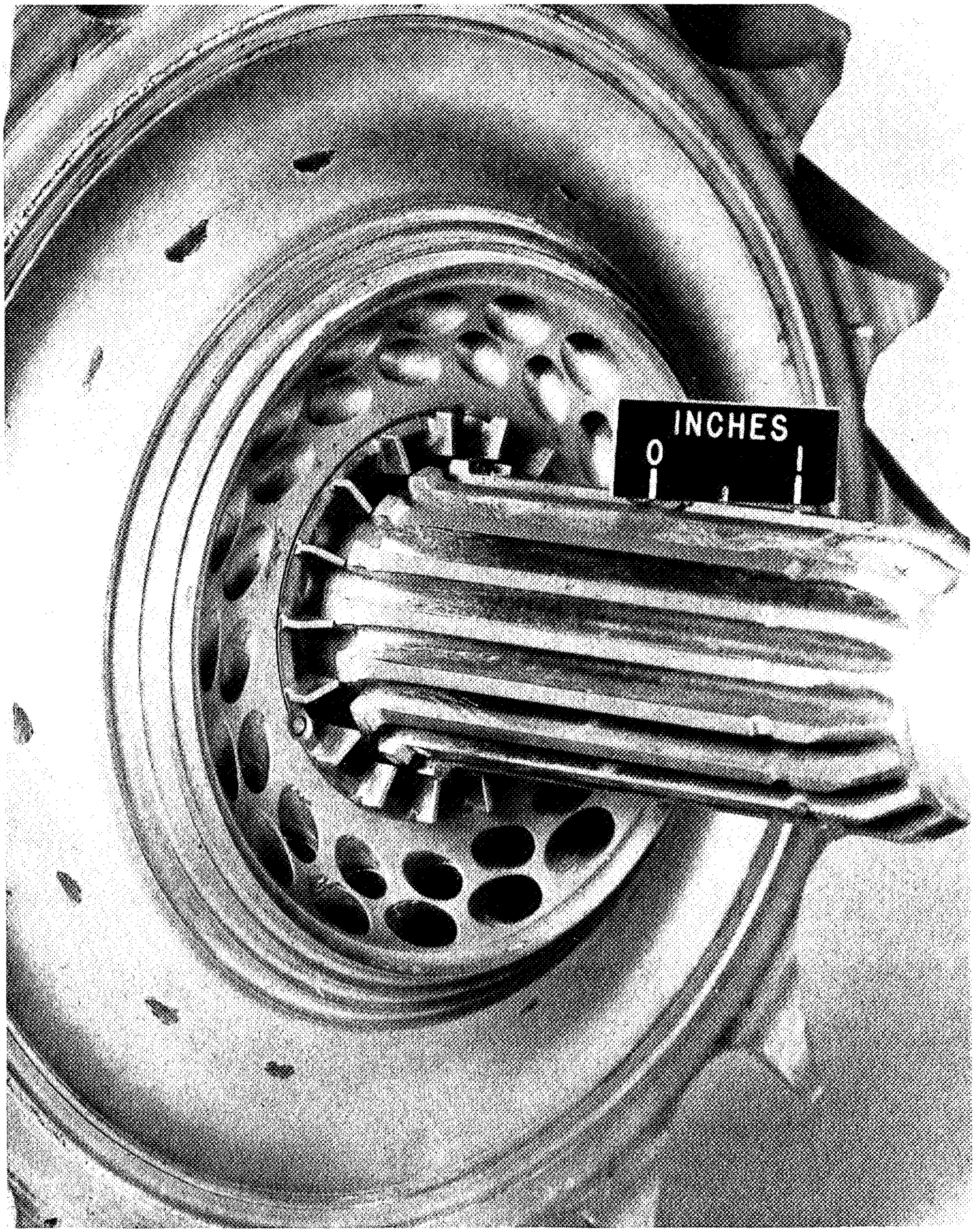

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Figure 1. - Shaft side of turbine rotor showing cooling-air inlet holes.

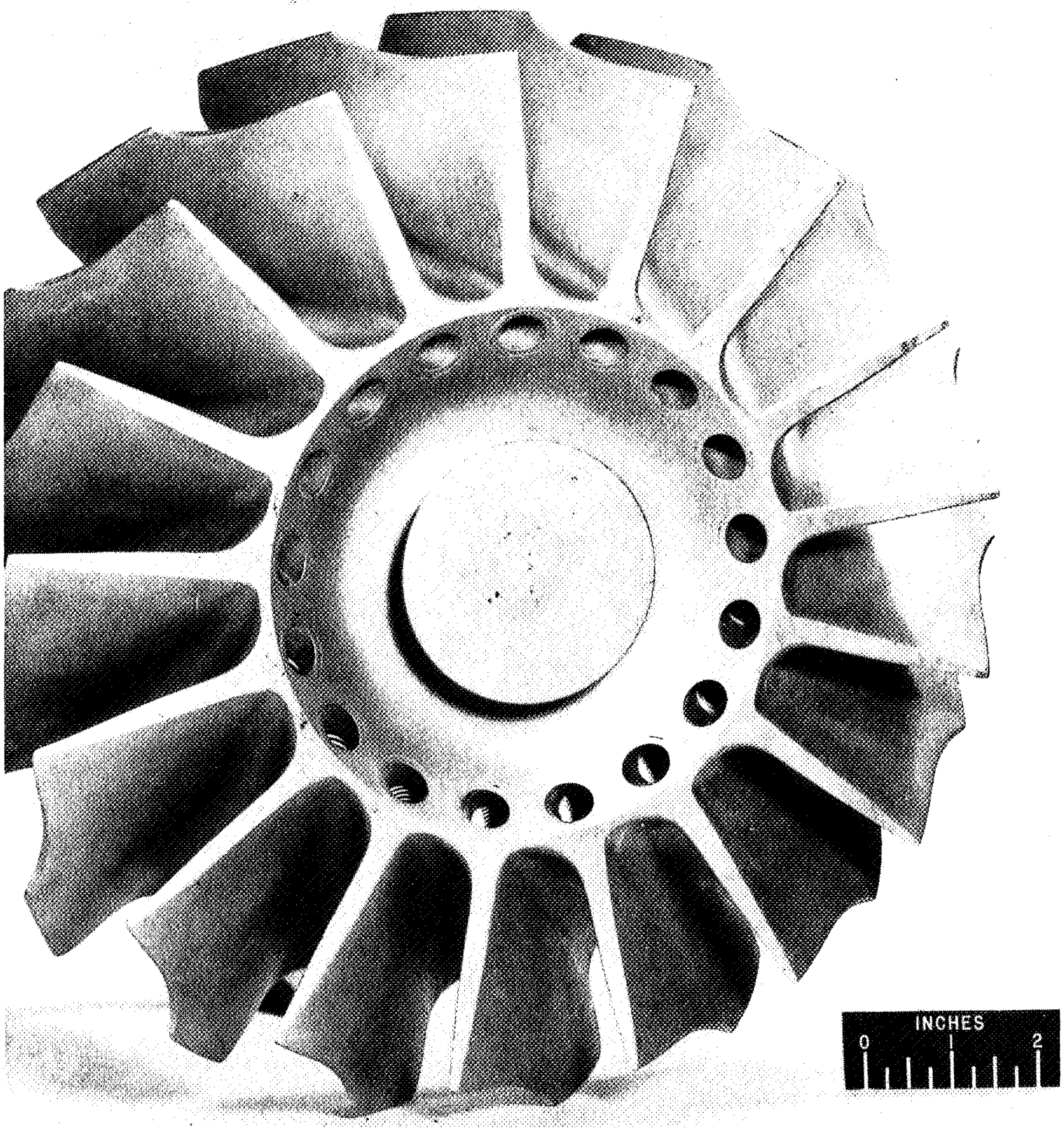

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Figure 2. - Downstream face of turbine rotor showing cooling-air outlet holes.

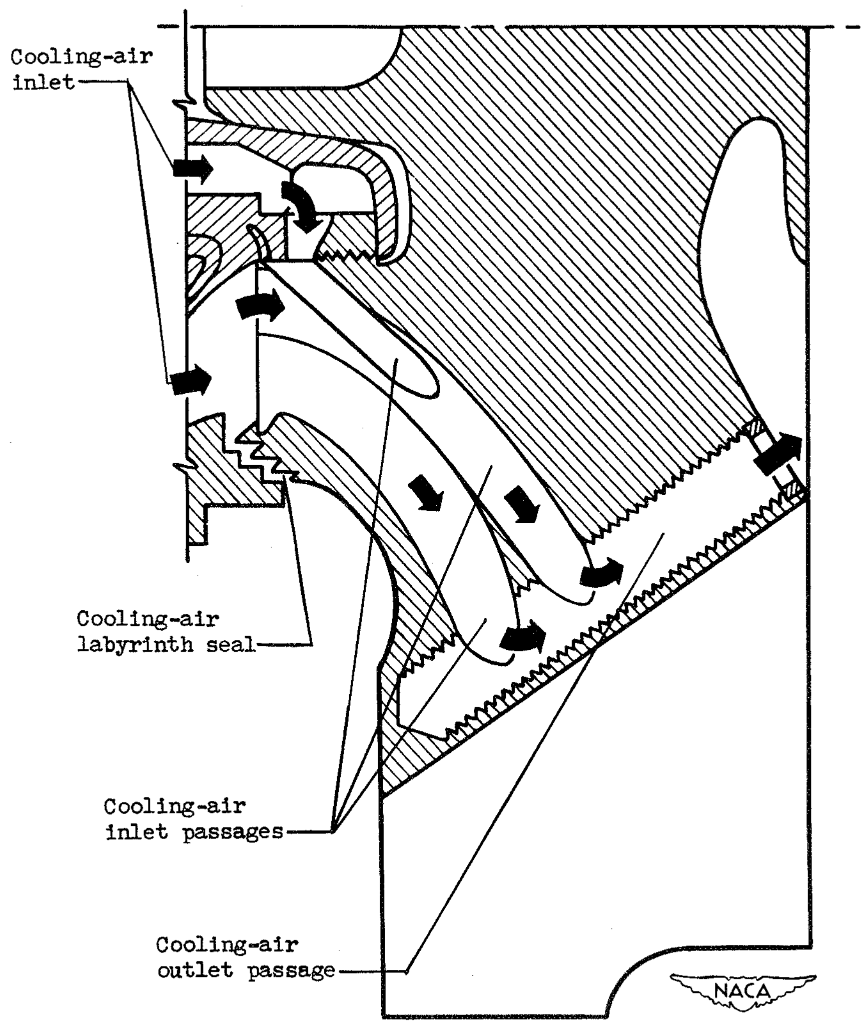

Figure 3. - Cross-section of radial-flow turbine wheel showing cooling-air passages.

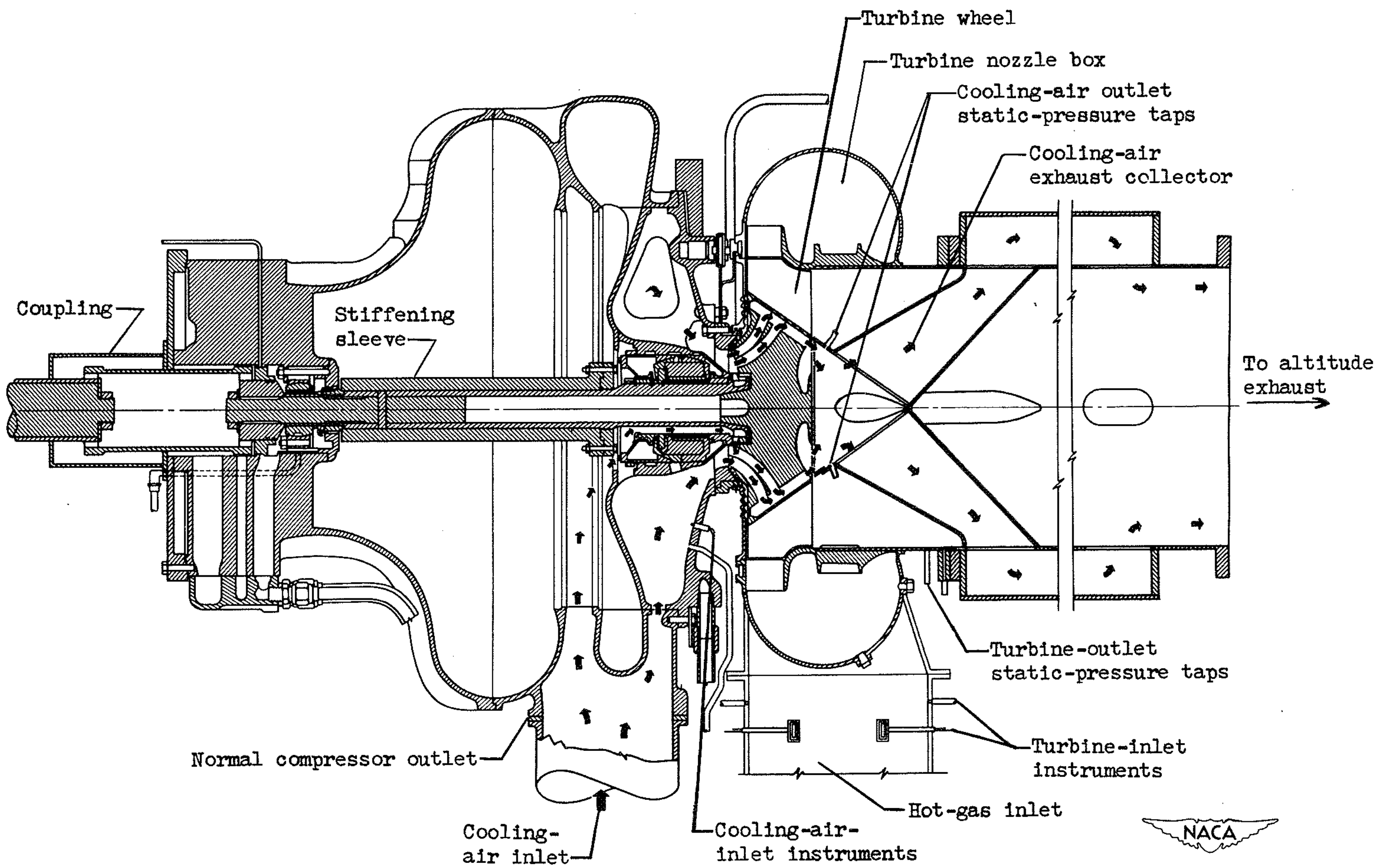

Figure 4. - Cross-section of setup for cooling investigation of radial-flow turbine showing instrumentation and cooling-air flow path.

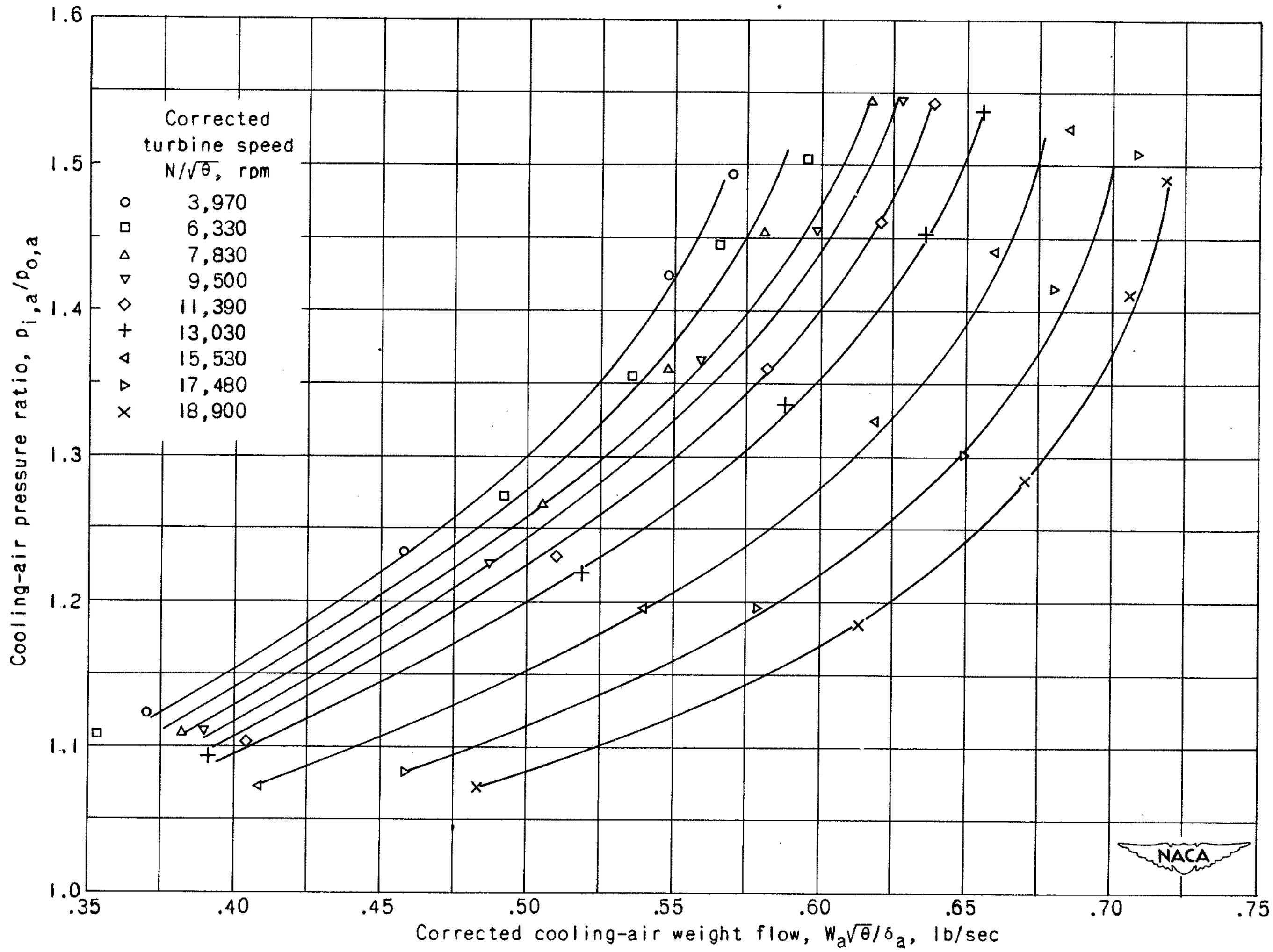

Figure 5. - Variation of corrected cooling-air weight flow with cooling-air pressure ratio for various corrected turbine speeds. Ratio of inlet total pressure to outlet static pressure, 2.0; inlet total pressure, 30 inches mercury absolute; inlet total temperature, 1600° R.

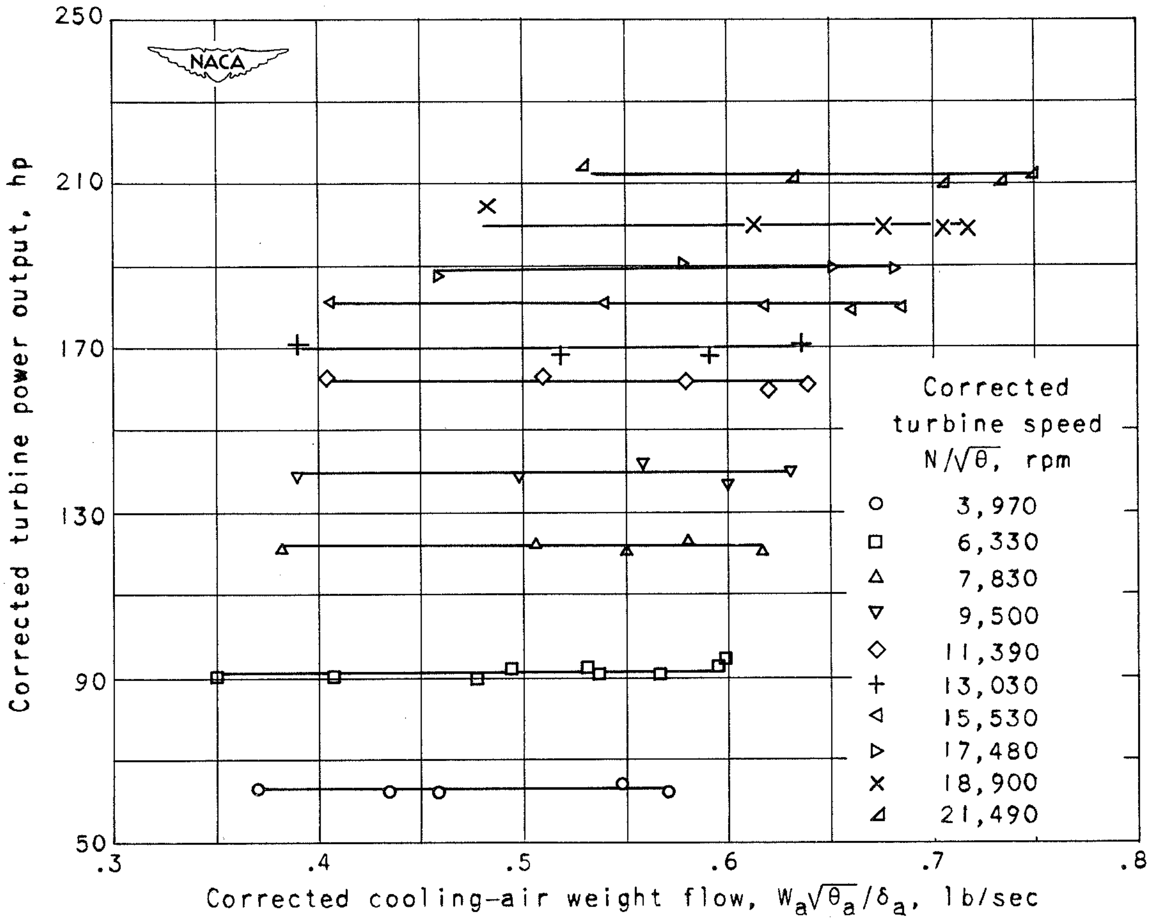

Figure 6. - Variation of corrected turbine shaft power with corrected cooling-air weight flow for various corrected turbine speeds. Ratio of inlet total pressure to outlet static pressure, 2.0; inlet total pressure, 30 inches mercury absolute; inlet total temperature, 1600° R.

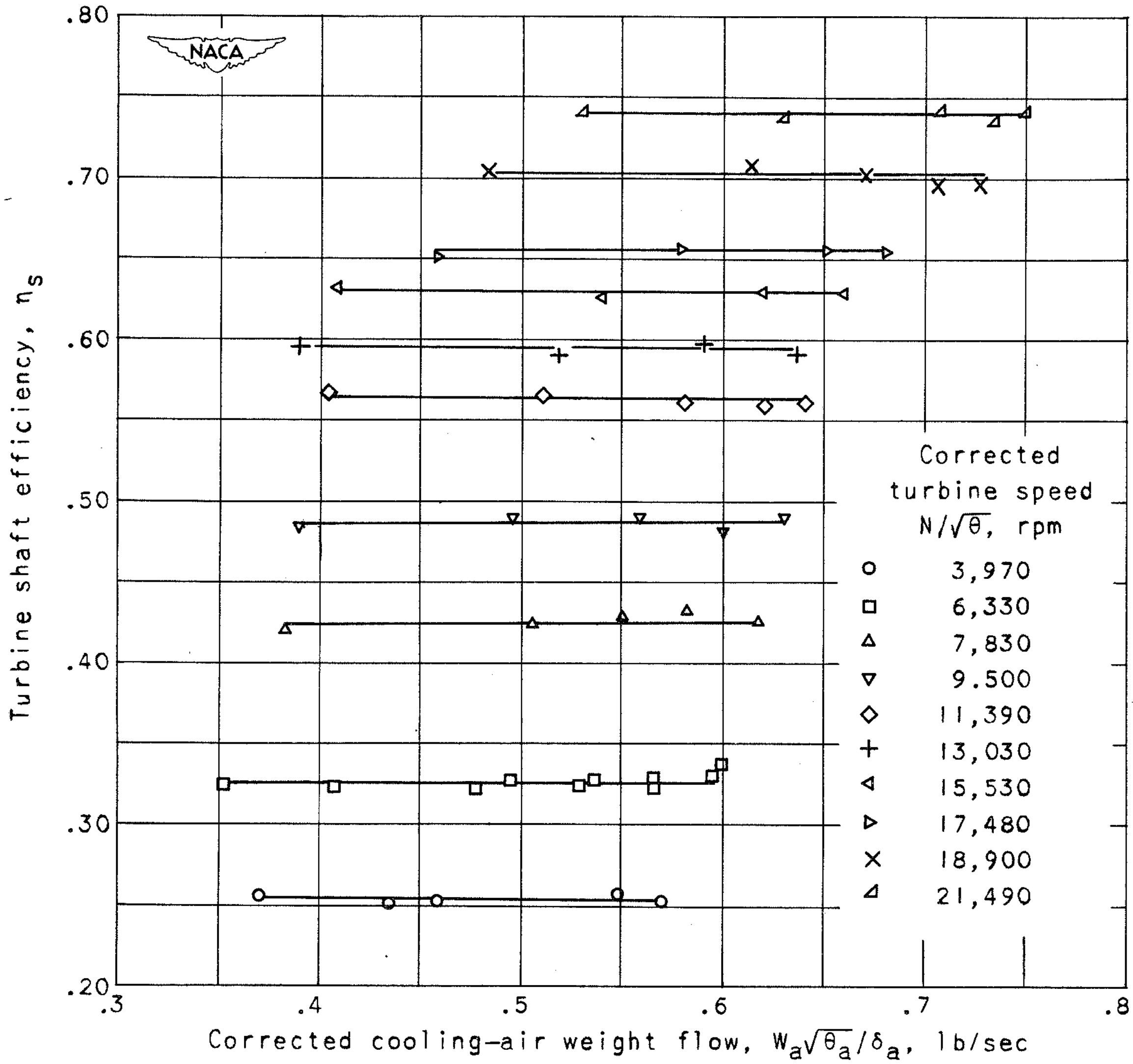

Figure 7. - Variation of turbine-shaft adiabatic efficiency with corrected cooling-air weight flow for various corrected turbine speeds. Ratio of inlet total pressure to outlet static pressure, 2.0; inlet total pressure, 30 inches mercury absolute; inlet total temperature, 1600° R.

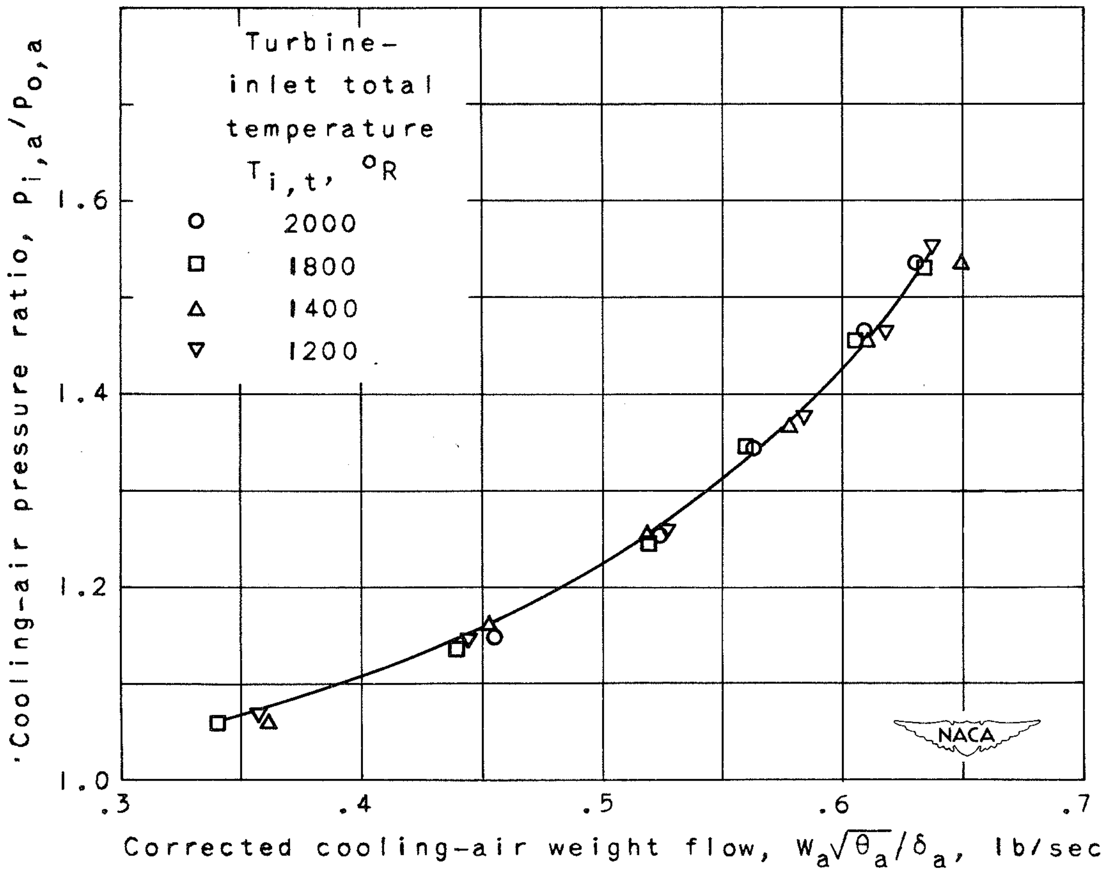

Figure 8. - Effect of turbine-inlet total temperature on corrected cooling-air weight flow at constant corrected turbine speed. Ratio of inlet total pressure to outlet static pressure, 2.0; inlet total pressure, 30 inches mercury absolute; corrected turbine speed, 12,000 rpm.

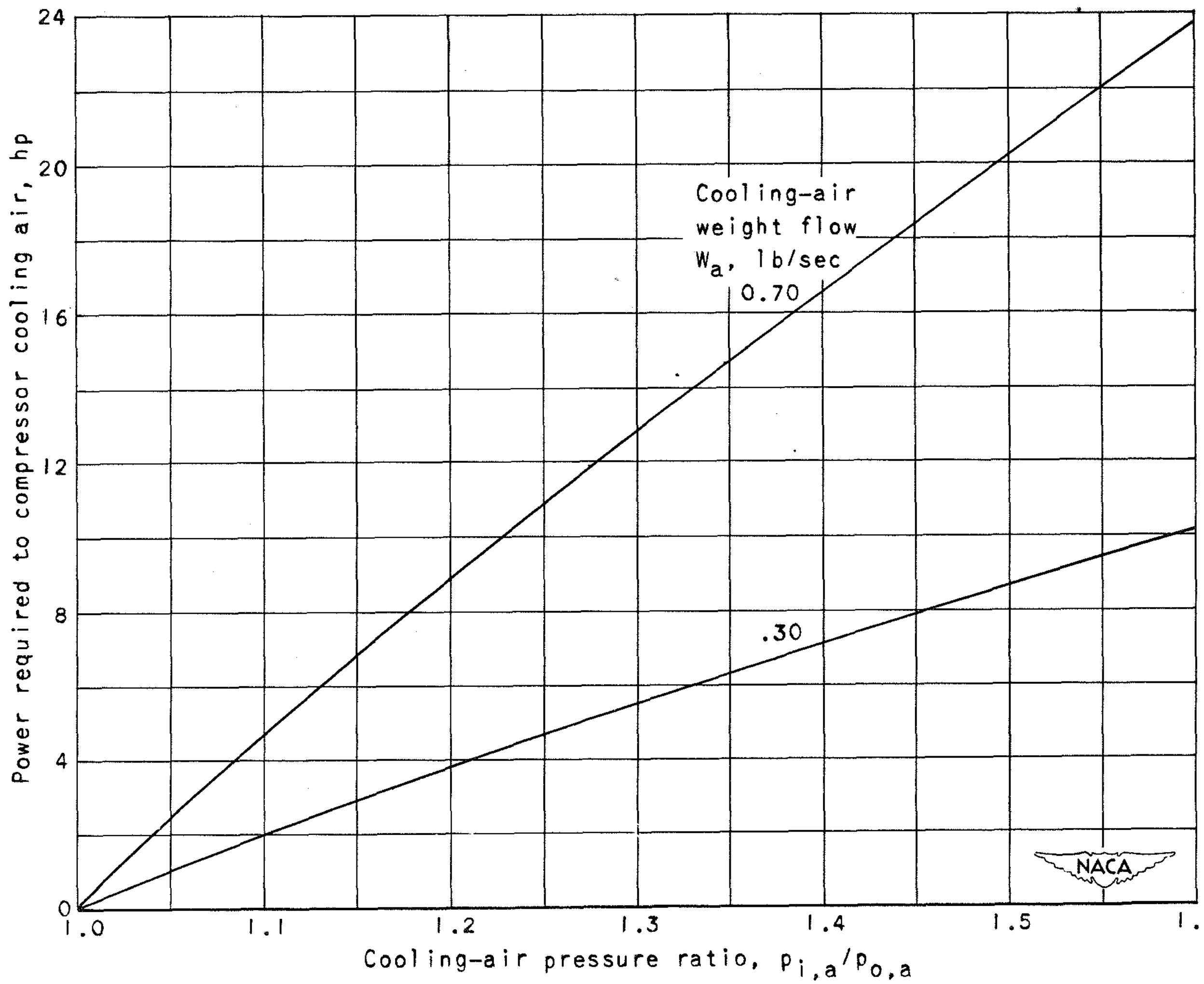

Figure 9. - Power required for compressing air for turbine-disk cooling in turbosupercharger unit. Compressor-inlet total temperature, 518.4° R; compressor efficiency, 0.75.